Torsional vibration measurements help in shaft crack diagnosis

by Agnes Muszynska, Ph.D.
Senior Research Scientist
Bently Rotor Dynamics Research Corporation
and Paul Goldman, Ph.D.
Research Scientist
Bently Rotor Dynamics Research Corporation

Rotor torsional vibrations are primarily measured only on machines which are subjected to considerable torsional excitations, such as synchronous and induction motor-driven machines, reciprocating machines, gear train drivers, and, generally, machines operating under variable torques. It is less well-known that rotor torsional vibrations may result not only from direct torsional excitations, but also from typical lateral excitations, such as unbalance, through which the torsional/lateral coupling mechanism is related to the eccentricity of the bending and twisting centers along the rotor.

Recent Bently Rotor Dynamics Research Corporation experimental and analytical research provides evidence that one important case of the torsional/lateral coupling mechanism is the radial constant load-resultant eccentricity of an anisotropic rotor. Anisotropic means having properties that are not the same in all radial directions.

Experimental results

Figure 1 illustrates the experimental test rig which is designed to model the effect of the radial load and shaft anisotropy on the torsional response. The rotor is driven by a synchronous one-half horsepower electric motor ►

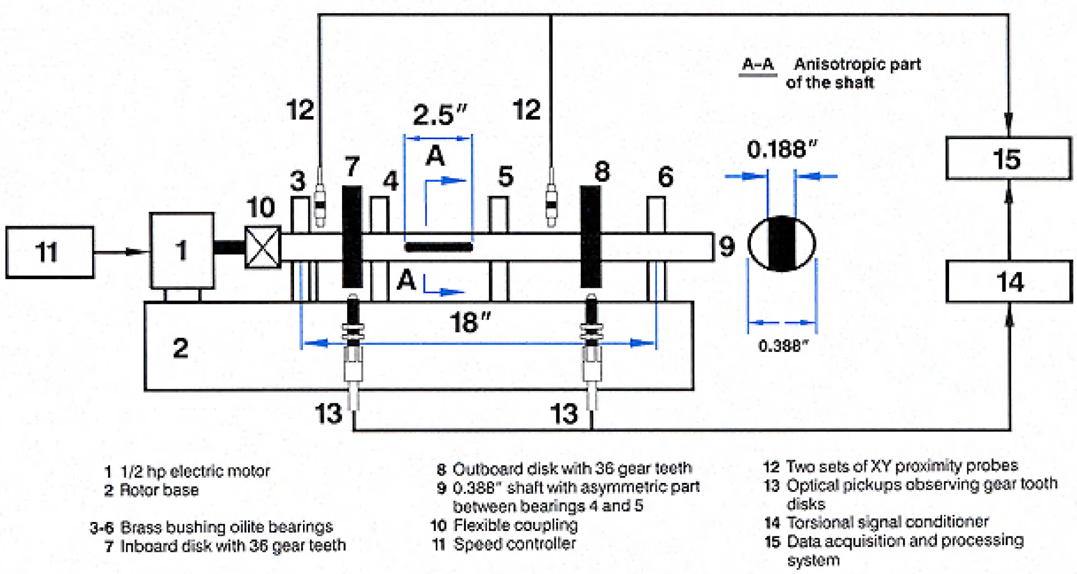

Figure 1
Experimental rotor rig.

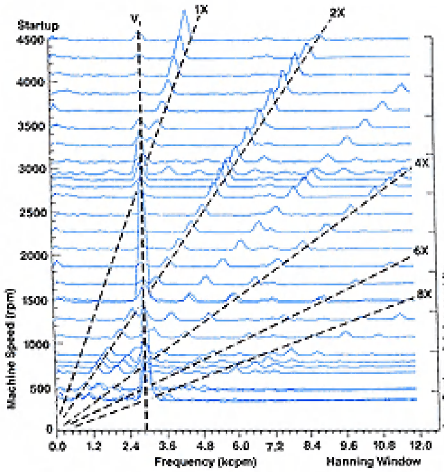

Figure 2
Spectrum Cascade of the rotor torsional response.

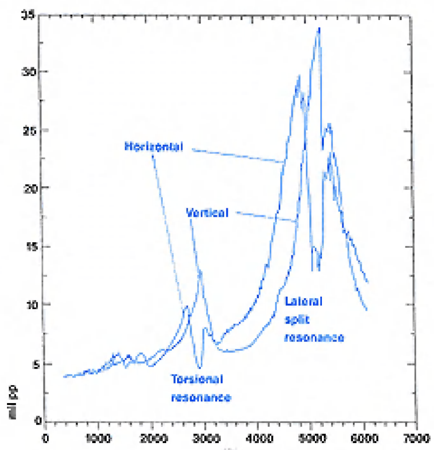

Figure 3
Rotor overall lateral vertical and horizontal vibration amplitude versus rotative speed at the outboard disk.

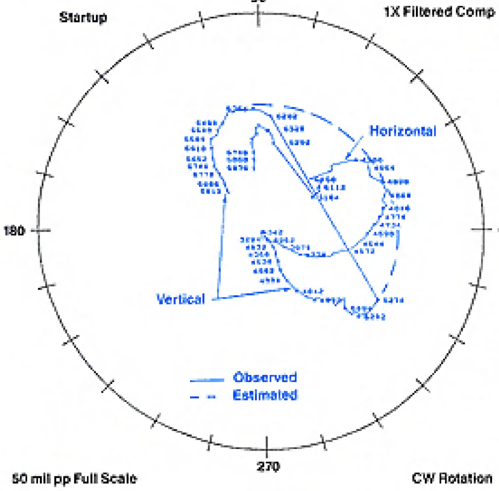

Figure 4
Polar plots of the rotor synchronous (1X) vertical and horizontal response at the outboard disk.

which is connected through a laterally-flexible, torsionally-rigid coupling. A speed controller controls the rotor rotative speed and acceleration. There are two disks fixed on the shaft, each with 36 precisely machined notches.

Optical pickups observe the disk notch passing frequency of both disks. In order to simulate anisotropy of the shaft, a part of the shaft between the two disks has been machined to produce two flats. The data is then processed through the Bently Nevada Torsional Vibration Signal Conditioner (TK17) to obtain a signal proportional to the angular displacement of both disks relative to the rotational angle of the rotating shaft. The conditioned signal of the inboard disk is then subtracted from the outboard disk signal to produce the dynamic twist (torsional vibrations) of the shaft between the two disks.

The shaft lateral vibrations are observed by two sets of two proximity probes in XY configuration at the inboard and outboard disk locations. In order to excite responses due to the anisotropy of the shaft, a shaft bow at the midspan was created by misaligning the shaft at the additional Bearings #4 and #5 using shims of 0.125 inches (3.175 mm) at their supports. The rotor was then balanced. When the shaft was rotating, vibrations occurred due to the interaction between the shaft bow-related side load and shaft anisotropy. The residual unbalance created an additional, minimal effect. Both lateral and torsional vibration data were recorded and reduced using Bently Nevada's ADRE®3 data acquisition and processing system.

The experimental results are presented in Figures 2 to 6. Due to the proximity of the relatively rigid Bearing #3, the shaft lateral displacements at the inboard disk location are very small. The lateral vibrations are presented by the data measured by the outboard disk probes. The results of torsional measurements exhibit the first torsional natural frequency, V_t, of 48 Hz (2880 rpm), which is excited at the rotative speeds of 360 rpm, 480 rpm, 720 rpm, 1440 rpm, and 2880 rpm (Figures 2 and 3). These speeds correspond to 8X, 6X, 4X, 2X, and 1X rotative speed excitation frequency.

The highest torsional resonant amplitudes occur at the rotative frequencies corresponding to 2X. All torsional response peaks are very high and sharp, which indicates that torsional damping is small. Direct overall lateral responses show several peaks: the first peak occurs around 3000 rpm which corresponds to the torsional natural frequency. The two highest peaks occur between 4700 rpm and 5500 rpm which corresponds to lateral resonances (Figure 4). Most of the lateral responses are due to the 1X vibrations (Figure 5). The highest peaks correspond to the split 1X resonance, which is due to the residual unbalance and the rotor/support anisotropy.

Figures 5 and 6 are Bode plots of filtered 1X and 2X torsional response components. These graphs exhibit both torsional and lateral mode peak amplitudes.

Conclusions

In the considered range of rotative speeds, the analysis of lateral vibrations alone does not provide sufficient information to diagnose the existence of the shaft anisotropy. A small amount of the 2X component present in the lateral vibrations Spectrum could be caused by the bow-related side load on the shaft alone. At the same time, the torsional response exhibits distinct high peak responses with the first torsional natural frequency at rotative speeds of 2880/2i (i = 1,2,3,4). These responses are caused mainly by the shaft anisotropy. (Rotor torsional responses taken before machining the flats on the shaft showed a very low level of vibration.) This strong effort, observed at low rotative speeds, should be used for shaft crack diagnostic purposes.

The torsional vibrations were characterized by very high amplitudes due to poor damping in the torsional modes. These torsional vibrations had a considerable magnitude at quite low rotative speeds. This suggests that it is very important to measure torsional vibrations on rotating machines, even though there are no evident torsional excitations and, thus, no torsional problems expected. Trended torsional vibrations may provide an invaluable warning signal about a propagating transversal crack on the rotor, as the crack causes an increase of rotor anisotropy.

Torsional vibrations are often considered beyond the scope of ordinary machinery monitoring and diagnostics on rotating machines. This situation may, however, soon change. With advancements in transducer design, the torsional vibrations monitored along with lateral vibrations create a new tool in machine health diagnostics which is especially useful in the early detection of shaft cracks. ■

Editor's Note: *This article is an excerpt from the paper, "Torsional/Lateral Vibration Cross-Coupled Responses Due to Shaft Anisotropy: a New Tool in Shaft Crack Detection," published in the Proceedings of the Conference on Vibrations in Rotating Machinery, Bath, United Kingdom, 1992, and reproduced here by permission of the Council of the Institution of Mechanical Engineers, London, England.*

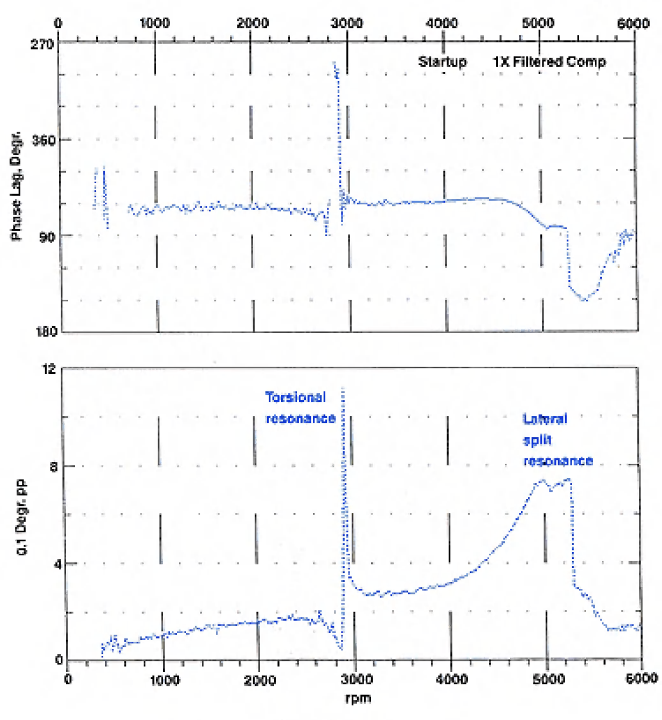

Figure 5
Bode plot of the rotor synchronous (1X) torsional response versus rotative speed. Estimated torsional first mode damping factor is 0.01.

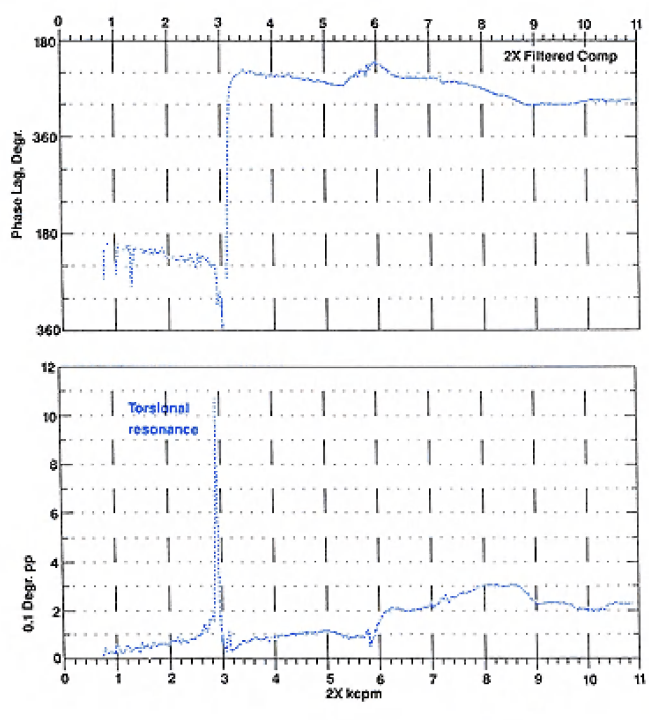

Figure 6
Bode plot of the rotor (2X) torsional response versus double rotative speed.